

Thermodynamic analysis and experimental investigation of a Solo V161 Stirling cogeneration unit

E.D. Rogdakis, G.D. Antonakos*, I.P. Koronaki

Laboratory of Applied Thermodynamics, School of Mechanical Engineering, Thermal Engineering Section, National Technical University of Athens, Heroon Polytechniou 9, Zografou Campus 15780, Athens, Greece

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ABSTRACT

In order to investigate the Stirling engine implementation technology, a Solo Stirling Engine V161 cogeneration module has been installed at the Laboratory of Applied Thermodynamics of National Technical University of Athens. A special thermodynamic analysis of the engine's performance has been conducted introducing and utilizing specially designed computing codes along with the thermal balance study of the unit. Measurements were conducted under different operational conditions concerning various heat load stages of the engine, working pressure, as well as electric power production. Analysis of the experimental results has shown that the overall performance of the Stirling unit proved very promising and quite adequate for various areal applications, equally competing with other CHP systems. The performance of the unit experienced significant stability all over the operating range. The power stand ratio 0.35 differentiates Stirling cogeneration units from others that use diverging technologies significantly. The energy savings using a Stirling CHP unit, in respect to the concurrent use of a thermal and an electrical system at the same equivalent power has revealed 36.8%.

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1. Introduction

Universally recognized as an effective solution to reducing CO₂ emissions, decentralized cogeneration systems are expected to become more and more prominent with time taking into consideration the policy choice for a 25% reduction in CO₂ emissions by 2025, which in combination with current conditions and projected needs will require the doubling of the electricity generated by cogeneration systems (CHP). Within this framework, great interest has been exhibited in low power systems (m-CHP) 1–10 kW_{el}, which are designed to cover equivalent heat loads, as reflected in the related literature which includes a significant number of comparative studies. One of these studies compares different cogeneration systems based on the complexity, range of power and efficiency they can achieve, as well as their ability to combine with other thermal systems in order to improve the overall efficiency of the facility [1]. In another work, selected cogeneration systems were compared on the technical, ecological and economic level and in terms of specific energy needs, and they have exhibited both very good performance and flexibility so as to gain access to the market

for domestic and small-scale industrial use [2]. Given the fact that the Stirling engine incorporates unique features which make it quite suitable for power production decentralization, the concept of using Stirling engines in general, and particularly in CHP systems, has consequently been brought up again, both as a study issue as well as a subject of research. Stirling engine CHP systems are now recognized for their efficiency and long service life, especially when compared to corresponding systems driven by internal combustion engines [3]. Research and development in this sector have led to improvements in the operational characteristics and performance of Stirling cogeneration systems. The design and construction of new burners using flameless combustion process (FLOX) have significantly reduced the Stirling CHP systems emissions compared to the corresponding pollutant emissions of conventional power systems [4,5]. In contrast to the Otto and Diesel engines, Stirling engines carry hermetically sealed cylinders which utilize all types of external heat sources independent of the type of fuel used. Heat can be supplied to a Stirling cylinder by burning diesel, gasoline, liquefied propane gas (LPG), liquefied natural gas (LNG), biomass, bio-fluids, or by directing concentrated solar irradiance. Solar energy is considered to be a promising source of heat energy in these systems, thus bringing them to the forefront of the global efforts for green development [6]. In addition, since the capacity of electric power production is an important selection factor for the appropriate cogeneration system, the fact that Stirling engines

* Corresponding author. Tel.: +30 2107724071; fax: +30 2107723670.

E-mail addresses: rogdemma@central.ntua.gr (E.D. Rogdakis), gantonak@central.ntua.gr, antonakosgeorge@yahoo.gr (G.D. Antonakos), koronaki@central.ntua.gr (I.P. Koronaki).

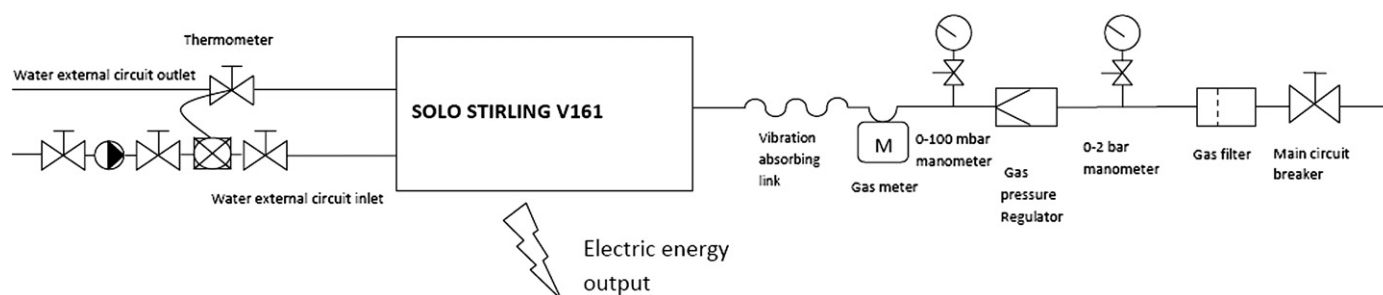


Fig. 1. Solo Stirling engine V161 installation.

provide the potential to produce adequate electrical power across the full range of the unit's load is of major significance.

2. Description of the unit

For academic and research purposes a Solo Stirling Engine V161 cogeneration module has been installed at the Laboratory of Applied Thermodynamics of the National Technical University of Athens [Fig. 1]. As already mentioned, the Stirling engine justifiably constitutes a promising option for both home and industrial applications especially in the field of pollutant emission reduction. In particular, the incoming heat is derived by an external continuous combustion burner at constant operating conditions which leads to complete burning resulting in the minimization of CO emissions. In addition, owing to the specially designed preheated burner with which the Solo Stirling 161V unit is equipped and which uses a flameless oxidation burner (FLOX), the formation of nitrogen oxide is reduced while the combustion efficiency substantially improves [4]. A detail Solo engine emission report has been published by Baumuller et al., according to which the exhaust emissions of NO_x, CO and HC respectively constitute 18%, 8.6% and 99% of the exhaust emissions of a small-scale CHP with catalyst (legal emission limit) [6]. Compared to a CHP with catalyst and lambda sensor, a Stirling engine the testes CHP unit exhibits a 33% reduction in NO_x emissions, and reductions of 15% and 90% in CO and HC emissions respectively.

As already mentioned, a Solo Stirling V161 cogeneration unit has been installed at NTUA with the entire installation procedure carried out according to the manufacturer's instructions [5]. The purpose of its installation and operation was to study the experimental results of the engine's thermodynamic performance, the analysis of which was made possible by relevant computer codes especially developed to track the facility's thermal balance study and to investigate the unit efficiency on a wide range of power outputs. The working gas mean pressure ranged from 30 to 130 bars while the working gas temperatures in the heater and cooler, controlled by the unit's electronics, were kept constant and equal to $T_h = 700^\circ\text{C}$ and $T_k = 30^\circ\text{C}$, respectively. The working gas mass in and out of the engine's working spaces was also controlled electronically in order to achieve the desired average operating pressure. The geometric and functional characteristics of the installed Stirling engine are shown in Tables 1 and 2, respectively, as provided by the manufacturer [5].

3. Thermodynamic analysis

The computing code developed in this study was based on the Stirling engine adiabatic thermodynamic analysis model, as presented by Urieli and Berchowitz [7].

In addition to the shown engine's geometric characteristics, mechanism features and components, the necessary module operating input parameters for running the code were: Engine's speed [Hz] or [rpm] Heater's temperature (constant) [K] Cooler's temperature (constant) [K] Working gas mass [kg]

The thermodynamic analysis applied to the Solo Stirling Engine V161 cogeneration unit was based on the ideal adiabatic model, as proposed by Urieli and Berchowitz [7].

The diagrams shown below were derived for an average pressure of working gas (He) equal to $p_{\text{mean}} = 80$ bar, with energy characteristics calculated for each crank angle degree. Fig. 2 shows the working gas pressure as a function of crank angle. The pressure was considered to be constant throughout all working spaces of the machine at crank angle (time).

The program runs 360 times for each cycle of operation. Due to the fact that the working gas reciprocates in the engine, it generates particularly difficult conditions for the rapid stabilization of the system. However, on the other hand, the small molecular weight of the working gas, coupled with the steady heat flux to the heater as well as the constant crank rotation rate, creates advantageous conditions for the steady energy state. The derivation of the temperature, the pressure and the mass of the working gas are recorded at each time step (crank degrees) until stable state condition is achieved. Fig. 2 presents the transition at steady state condition of the working gas pressure while Fig. 3 presents the pressure transition to steady state. Figs. 4 and 5 present the transition at steady state condition of the working gas temperature and mass respectively. It takes less than 10 cycles for the steady state conditions to be established.

As observed in Fig. 4, T_c and T_e were affected by the cooler T_k and the heater T_h temperatures, respectively. Higher T_k and T_h required more cycles for establishing steady state conditions. It was also found that temperature T_e could exceed the heater metal wall constant temperature T_h , and that temperature T_c could be less than the cooler metal wall temperature T_k , which could be explained by compression and expansion procedures in the adjacent cylinders.

Fig. 6 shows the gas mass flow into the heater g_{Ah} , the regenerator g_{Ar} and the cooler g_{Ak} . The positive or negative shown mass flow is defined by the gas flow direction. As "positive" or "cold" was

Table 1
Geometric characteristics of Solo Stirling engine V161.

Engine's data	
Type	Solo Stirling 161
Plant	V – 2 Stirling (α type)
Cylinder's swept volume	160 mm ³
Working gas	He
Working gas maximum pressure	150 bar
Speed	1500 rpm

Table 2
Functional characteristics of Solo Stirling engine V161.

Burner and combustion chamber	
Burner's effectiveness	16–40 kW
Fuel	Natural gas
Gas components	B 23
Gas line pressure	50 + 15/–5 kPa
Exhaust gas back pressure	Max. 2 kPa
Exhaust gas temperature (max)	85 °C
Burner's system	Flameless oxidation
Flame control system (start–operation)	Oxidation – temperature
NO _x	80–120 mg/m ³
CO ₂	40–60 mg/m ³
Condensate (max load)	Approx. $1.667 \cdot 10^{-5}$ m ³ /sec

considered the flow from the compression space to the expansion space, corresponding to crank position from 0° up to 180. Similarly, for crank position from 181 to 360°, the flow was reversed, so it was referred to as “negative” or “hot”. According to the results, the “cold” mass flow at the interfaces between the heat exchangers was smaller, in absolute terms, which proved that a significant portion of the working gas mass remained continuously in the cooler volume, and could, therefore, be classified as “inactive mass”. As a result, the heat transferred and thus, the efficiency of the engine were both reduced. Similar conclusions were reached for the regenerator and for the heater volumes. The transferred thermal energy and the power output are shown in Fig. 7. The heat energy transferred from the working gas into the regenerator matrix and stored there in the hot flow, was fully recovered to the gas during the cold flow, which is consistent with an ideal regenerator consideration.

As observed, the net amount of the heat which remained in the heater was transferred to the working gas at the end of the cycle. At the same time, heat was ejected by the cooler. Finally, according to Fig. 7, it was realized that work produced during half of the crank rotation (from 134 up to 314°).

Fig. 8 shows the pressure–compression space volume diagram ($p-V_c$), the pressure–expansion space volume diagram ($p-V_e$) and the pressure–total space volume diagram, clearance spaces included ($p-V$).

In Table 3, the results according to the adiabatic ideal gas analysis are presented.

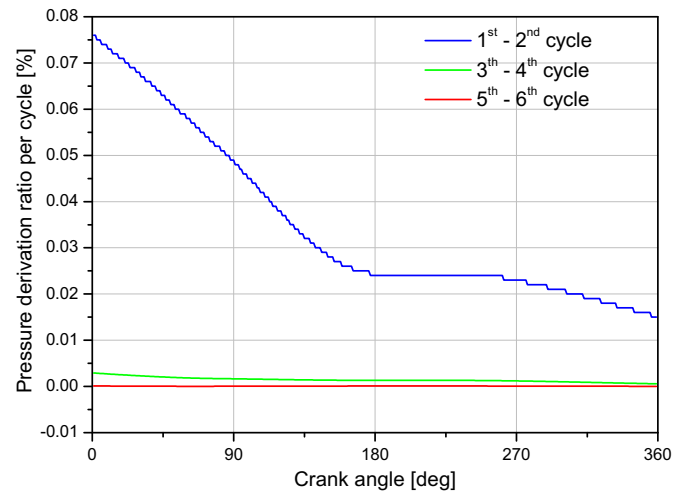


Fig. 3. Pressure derivation ratio. Transition to steady conditions.

3.1. Sensitivity analysis

A sensitivity analysis was performed on the inlet and outlet variables of the thermodynamic analysis of the Stirling engine in order to predict the effect of the inlet variables on the temporal and spatial performance of the engine. Independent variables constitute the inputs: Engine's speed [rpm], Heater's temperature (constant) [K], Cooler's temperature (constant) [K] and the Mean Pressure [kPa]; dependent variables constitute the outputs: Mean Regen press Drop [kPa], Mean Energy Dissipation [W], Regenerator's eff. [-], efficiency [-], Power [kW], Work [J/cycle], Heat input [J/cycle], Heat output [J/cycle]. A regression analysis has also been conducted with the view to investigate the degree of influence of each independent variable (quantitative or not) on the dependent variables. Table 4 shows the statistic results, after running a multiple regression on the predicted Heat Output treated as a dependent (Indicator) variable next to the independent variables.

The standardized regression coefficients indicate the degree of influence of every input, regardless of sign. As observed, the larger the coefficient of the independent variable is, the greater the influence is on the dependent variable, whereas a negative

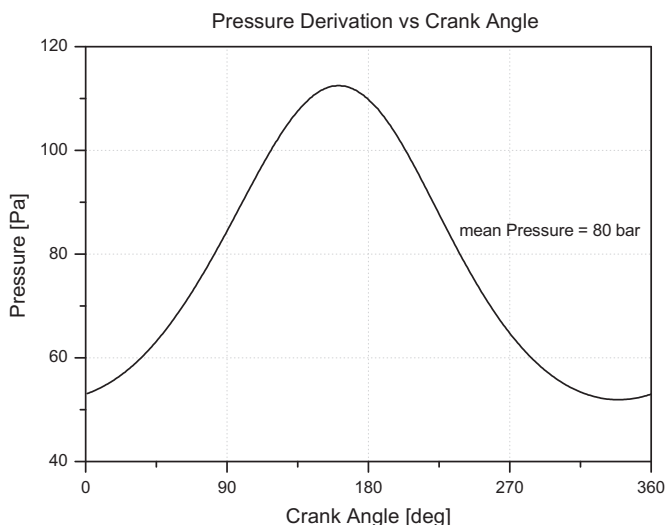


Fig. 2. Working gas pressure per crank angle.

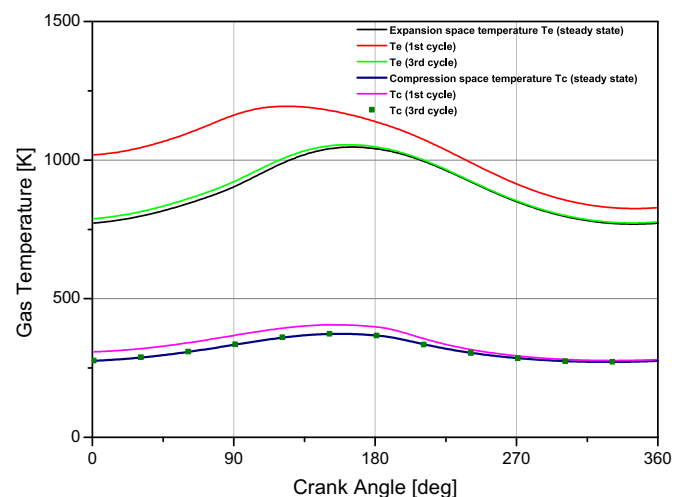


Fig. 4. Compression space temperature T_c and expansion space temperature T_e per crank angle degree. Transition to steady conditions.

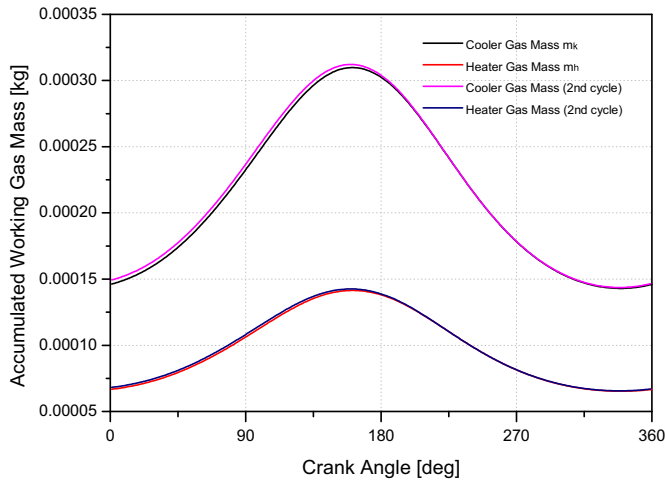


Fig. 5. Accumulated gas mass in the cooler and heater per crank angle degree. Transition to steady conditions.

coefficient indicates that the dependent variable decreases when the corresponding independent variable increases. The linear regression equations for the dependent variables are shown in Table 5.

Considering the results of the sensitivity analysis, the following remarks can be made:

R-Square and adjusted R-Square which represent the quality of the linear fit were found greater than 86.69% and less than 99.99% (Table 5), for each output variable.

Since the input variables have different measurement scales, the regression coefficient of each input variable found at the end of a regression analysis was transformed to a standardized regression coefficient in order to compare the relative influence of the variable.

The standardized regression coefficients of the input variables, in descending influence, are: Engine's speed (-0.18% – 96.49%), Mean Pressure (11.89% – 100%), Cooler Temperature (-11.7% – 0.012%), Heater Temperature (-27.027% – 0.8295%).

The results of the sensitivity analysis show that the output conditions are found to be most sensitive to the engine's speed and the mean pressure, but least sensitive to the cooler temperature.

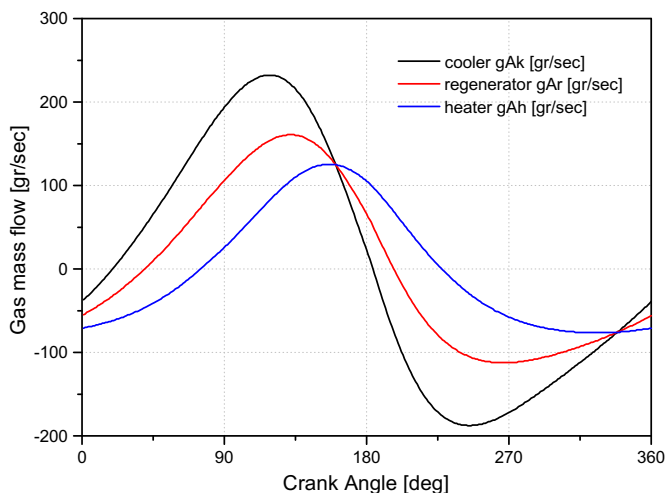


Fig. 6. Working gas mass flow into the Stirling engines heat exchangers per crank angle degree.

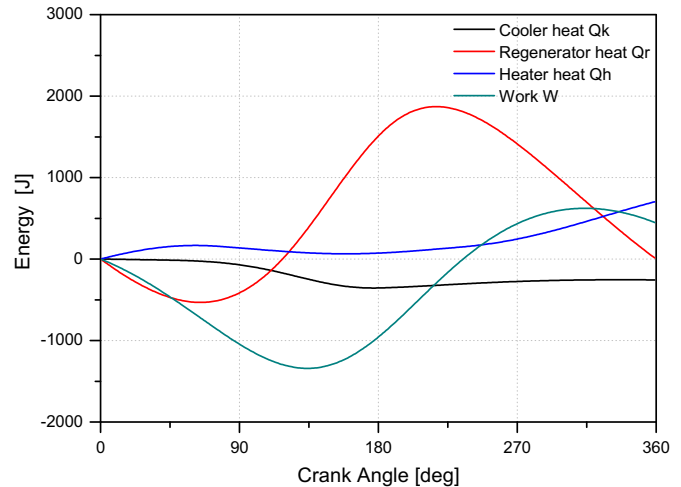


Fig. 7. Heat transferred energy and produced work per crank angle degree.

4. Solo Stirling engine v161 unit heat balance analysis

In order to calculate the heat balance of the system's cogeneration unit, the following indication and recording instruments were used:

- Thermometer Siemens WFZ.E110-I.
- Analog natural gas flow meter.
- Digital gas analyzer TESTO 327-1.
- Digital air velocity meter TESTO 425.
- Solo unit on board monitor.

The present Stirling engine unit used natural gas as fuel for its operation. The thermal energy which flew through the engine's parts, pipes and lubricant oil was finally rejected as radiant heat from the external surfaces and heat transferred by convection Q_a and picked up by the forced air flowing inside the cover shell of the unit. It was this air-stream with these specific thermo physical properties which was led to feed the combustion mixture system. The remaining air mass was rejected, forced out to the environment, by means of a fan, through a small hole at the rear panel of the insulated unit cover shell [8]. The electric power P_e was produced by an electric generator directly connected to the

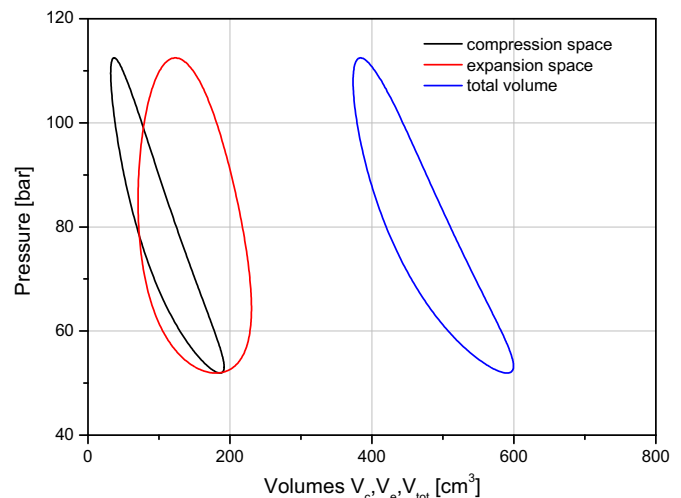


Fig. 8. Pressure P vs. space volume diagrams, for $p_{mean} = 80$ bar.

Table 3

Ideal adiabatic thermodynamic analysis results for the Solo Stirling V161 cogeneration unit.

P_{mean} [$\times 10^5$ pa]	30	40	50	60	70	80	90	110	110	120	130
Q_k [J/cycle]	−101.114	−134.791	−168.427	−202.104	−235.781	−269.475	−303.177	−336.854	−370.531	−404.208	−437.885
Q_r [J/cycle]	0.229	0.305	0.382	0.458	0.534	0.611	0.687	0.763	0.84	0.916	0.992
Q_h [J/cycle]	266.94	355.848	444.646	533.553	622.461	711.412	800.384	889.292	978.199	1070	1160
W [J/cycle]	162.465	216.575	270.62	324.73	378.841	432.978	487.128	541.239	595.35	649.46	703.571
W_c [J/cycle]	−103.043	−137.362	−171.639	−205.959	−240.278	−274.615	−308.959	−343.279	−377.598	−411.918	−446.237
W_e [J/cycle]	265.507	353.938	442.259	530.689	619.119	707.593	796.088	884.518	972.948	1060	1150
P [kW], ($f = 25$ Hz)	8.12	10.8	13.5	16.2	18.9	21.7	24.4	27.1	29.8	32.5	35.2

Table 4

Sensitivity analysis of Heat Output. Treated as dependent qualitative (dummy) variable.

Heat output [J/cycle]						
Regression statistics			Regression coefficients	Standardized regression coefficients	Standard error	t-Stat
Multiple R	0.9999930	Intersept	−152.91400		32.29845	−4.7344
R-Square	0.9999860	Engine's Speed [RPM]	−0.00069	−0.00239	9.28200E-05	−7.43091
Adjusted R-square	0.9999850	Mean Pressure [kPa]	−0.03186	−0.99681	1.03200E-05	−3087.8
Standard Error	0.3726220	Cooler Temperature [K]	−0.11365	−0.00118	0.05676	−2.00216
		Heater temperature [K]	0.19786	0.00830	0.015029	13.16554
						1.988E-27

engine's crankshaft, while its value was indicated by the unit on the board monitor.

4.1. Energy offered to the unit. Q_u

The energy of the incoming mixture was apparent by the calorific capacity of the fuel, the temperature of the incoming air and the mixture ratio. Data about the composition and calorific capacity H_u of the natural gas used were given by the gas provider company. Gas consumption was measured by an analog natural gas flow meter, while the temperature of the incoming air was measured by a digital air thermometer. The mixture ratio was measured by the digital gas analyzer with the energy of the incoming mixture converted to heat inside the combustion chamber. The combustion efficiency PENDING was measured by a digital gas analyzer.

4.2. Rejected heat to the external cooling circuit. Q_w

The excess of heat was transferred through the cooler to an internal cooling circuit. The cooling water of the internal cooling

circuit passed through a gas-water heat exchanger so that heat was recovered from the exhaust gases. Thereafter, the heat Q_w absorbed by the internal cooling water circuit was transferred, via a plate heat exchanger, to an external water heating circuit providing hot water for any use.

4.3. Flue gas heat waste. Q_c

The heat Q_c which remained in the waste gases was rejected into the environment with maximum gas temperature reaching 85 °C. In order to calculate the flue gas heat losses Q_c , the temperature, volumetric rate, as well as the composition of the exhaust gas determined by the digital gas analyzer need to be known.

4.4. Unit ventilation heat losses. Q_a

The thermal energy which flew through the engine's parts and pipes and was carried by the lubricant oil was finally rejected by radiation from the external surfaces and by convection Q_a due to the forced air flowing under the cover shell of the unit. In order to calculate the ventilation heat losses Q_a , the air temperature and air velocity need to be measured, using a digital temperature/velocity meter.

4.5. Produced electrical output. P_e

Electric power P_e was produced by the generator directly connected to the crankshaft and its value was received and indicated on the unit's board monitor. The supply of the local electricity network to the Solo Stirling Engine V161 became stable and feasible when the engine reached the equilibrium state at the speed of just over 1500 rpm (Table 8).

4.6. Results

Results of the heat balance analysis of the CHP Solo Stirling V161 unit covering the entire load range of the engine are presented in Table 6. The percentage distribution of the total outcoming power to the unit is presented in Table 7. The electric efficiency of the unit (approx. 20%) showed remarkable stability throughout the operating range of the unit, indicating that the Stirling cogeneration unit

Table 5

Linear regression equations for calculating the output variables of the Stirling engine.

$Y = a + b \cdot \text{Engine's speed [rpm]} + c \cdot \text{Mean pressure [kPa]} + d \cdot \text{Cooler temperature [K]} + e \cdot \text{Heater temperature [K]}$					
Y	a	b	c	d	e
Mean regen press drop [kPa]	174.7750	0.01514	0.000207	−0.19202	−0.11683
Mean energy dissipation [W]	17425.4500	0.836633	0.010809	−20.6907	−11.3324
Regenerator's eff. [%]	1.0429	$−1.4 \cdot 10^{-6}$	$−2 \cdot 10^{-5}$	$−9.7 \cdot 10^{-5}$	$−1.1 \cdot 10^{-5}$
Efficiency [−]	0.7347	$−3.8 \cdot 10^{-5}$	$3.2 \cdot 10^{-6}$	0.00075	$7.68 \cdot 10^{-5}$
Power [kW]	307.4019	0.004849	0.000725	−0.33095	−0.20741
Work [J/cycle]	−854.0730	−0.02206	0.046197	0.173858	0.830974
Heat input [J/cycle]	−275.9720	−0.00132	0.081137	−0.19341	0.353114
Heat output [J/cycle]	−152.9140	−0.00069	−0.03186	−0.11365	0.197864

Table 6

Heat balance analysis results for the Solo Stirling V161 cogeneration unit.

$P_{\text{mean}} [\times 10^5 \text{ pa}]$	30	50	70	80	90	110	120	130
Q_u [kW]	12.6957	18.87052	22.8524	26.5456	29.04848	36.08	39.056	41.046
Q_f [kW]	0.050783	0.003223	0.15997	0.212365	0.32	0.4329	0.54679	0.6977
W_{el} [kW]	1.6	3	4.6	5.3	5.9	7.4	7.9	8.5
Q_w [kW]	8.8	12.04	14	16.3	18.5	22.2	23.83	24.2
Q_a [kW]	0.6446	0.8904	0.784001	0.8363	0.76258	0.7981	0.6447	0.752
Q_c [kW]	0.534619	0.72012	1.288	1.1859	1.4	1.7605	1.6323	1.7977
Q_{loss} [kW]	1.0657	2.1067	2.02	2.7153	2.2015	3.4873	4.506	5.09858
Heat in [J/cycle]	267.15	429.42	591.7	672.83	753.97	916.24	997.38	1078.52
Heat out [J/cycle]	−82.106	−145.83	−209.55	−241.41	−273.27	−336.97	−368.85	−400.71
Power divergence	3.44%	3.44%	−0.94%	−0.34%	1.67%	0.76	3.45%	4.60%
Heat in divergence	6.42%	6.85%	7.05%	9.34%	7.82%	6.14%	7.76%	11.91%
Heat out divergence	1.97%	4.00%	6.94%	10.93%	9.25%	12.16%	12.43%	11.41%

Table 7

Distribution of the outcoming power from the Solo Stirling V161 cogeneration unit.

$P_{\text{mean}} [\times 10^5 \text{ bar}]$	30	50	70	80	90	110	120	130
Load [%]	23.08	38.46	53.85	61.54	69.23	84.62	92.31	100.00
Q_f [% Load]	0.40	0.60	0.70	0.80	1.10	1.20	1.40	1.70
W_{el} [% Load]	12.60	15.90	20.13	19.97	20.29	20.51	20.23	20.71
Q_w [% Load]	69.31	63.80	61.26	61.40	63.61	61.53	61.00	58.96
Q_a [% Load]	5.08	4.72	3.43	3.15	2.62	2.21	1.65	1.83
Q_c [% Load]	4.21	3.82	5.64	4.45	4.82	4.88	4.18	4.38
Q_{loss} [% Load]	8.39	11.16	8.84	10.23	7.57	9.67	11.54	12.42
n_e [%]	12.60	15.90	20.13	19.97	20.31	20.51	20.23	20.71
n_{th} [%]	69.31	63.80	61.26	61.40	63.69	61.53	61.01	58.96
n [%]	91.61	88.84	91.16	89.77	92.42	90.33	88.46	87.58

was quite reliable in this rpm area. The total efficiency of the unit displayed great stability too, approx. 90%, throughout the load range (Fig. 9). The deviation to the thermal balance which was noticed to be about 10% – a very encouraging value – was caused by:

- incomplete thermal equilibrium achievement,
- random and systematic errors of the measuring instruments and the measurement procedure,

inconsistency of the gas composition, since it is a mixture of Russian and Algerian natural gas.

5. Solo Stirling V161 unit efficiency

The following energy factors were defined for cogeneration units [8]:

$$\text{Electrical efficiency} \quad n_e = \frac{P_e}{Q_u} \quad (1)$$

$$\text{Thermal efficiency} \quad n_h = \frac{Q_w}{Q_u} \quad (2)$$

Table 8

Electrical to thermal power standard ratio.

Unit type	Electrical to thermal power standard ratio (C) [9]
Combined gas turbine cycle with energy recovery	0.95
Backpressure steam turbine	0.45
Condensation steam turbine	0.45
Gas turbine with heat recovery	0.55
Internal combustion reciprocating engines	0.75
Solo Stirling V161 unit (with heat recovery)	0.35 (present study)

$$\text{Overall efficiency} \quad n = n_e + n_h \quad (3)$$

The electrical to thermal standard energy ratio of the Solo Stirling V161 unit is shown in Table 9, compared to the other CHP systems.

Fig. 9 presents the analysis of the experimental results.

In the sensitivity analysis, the theoretical analysis model results are studied in respect to the corresponding experimental ones. Figs. 10–14 shows the variations displayed relating to the incoming heat, the outgoing heat, the power, the performance and the power stand ratio respectively.

In order to fuel the cogeneration unit, fuel of calorific capacity Q_u is used. However, by means of the heater, heat Q_{in} enters the Stirling engine. The deviation between the modeled and experimental results does not exceed 10% (Fig. 10 and Table 6), which may be attributed to the fact that the equations for the dimensionless Nusselt number may require improvements in cases of reciprocating movement of the working gas. Additional heat Q_{out} is rejected by the Stirling engine via the cooler to an internal cooling circuit and then, by means of a plate heat exchanger, to the external cooling circuit. However, the heat recorded at this part of the unit (Q_w) also comprises of the heat absorbed by the lubricant as well as the rejected flue gas through the shell and tube heat exchanger. The deviation between the recorded heat which was rejected by the engine cooler and the respective value calculated by the model does not exceed 13% and is illustrated in Fig. 11.

Power output P , as calculated by the model, corresponds to the power output of the Stirling engine crankshaft. The crankshaft is

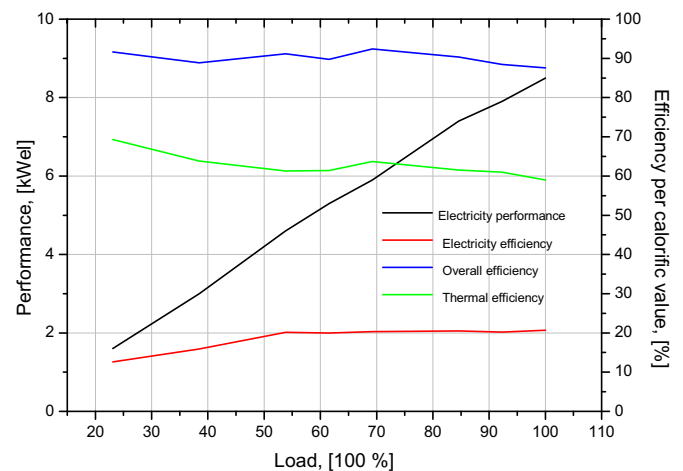
**Fig. 9.** Electrical performance and thermal efficiency of Solo Stirling V161.

Table 9

Internal combustion and Stirling engine cogeneration systems performance comparison.

	Internal combustion engines [3]		Stirling engines [3]			Solo Stirling V161 (present study)					
Capacity											
Electrical [kW _{el}]	1.0	5.0	0.8	3.0	9.5	1.6	3.0	4.6	5.3	5.9	7.9
Thermal [kW]	3.3	12.6	8.0	15.0	26.0	8.8	12.04	14	16.3	18.5	23.83
Efficiency											
Electrical [%]	20	25	10	15	24	12.60	20.13	20.13	19.97	20.31	20.23
Thermal [%]	65	63	75	75	72	69.31	61.26	61.26	61.40	63.69	61.01

directly coupled to the motor shaft, thus theoretically converting the total power output to electrical power W_{el} . Fig. 12 presents the association between the calculated power output and the power produced as recorded by the incorporated unit instruments. The deviation in the power output presented in Table 6 does not exceed 5%, increasing with any increase in the unit load, where motor slips or cases of working gas compressibility may occur, which absorb additional energy.

Figs. 13 and 14 present the correlation of the modeled to the experimental results of the engine performance and power stand ratio respectively. In terms of the engine performance and for the entire measurement range, the experimental and modeled results exhibit deviations of significantly minor variance in comparison with power and heat results. The modeled engine performance is constantly calculated to be higher than the respective experimental performance, with an average deviation of 10%. This deviation could be attributed to convective heat transfer phenomena, shuttle losses, the operation of the buffer space and the probable non-adiabatic behavior of the regenerator, and should be further examined in the future. Moreover, the state of thermal equilibrium may not have been established either, despite the fact that the thermocouples with which the unit is equipped recorded steady temperatures at different parts of the engine.

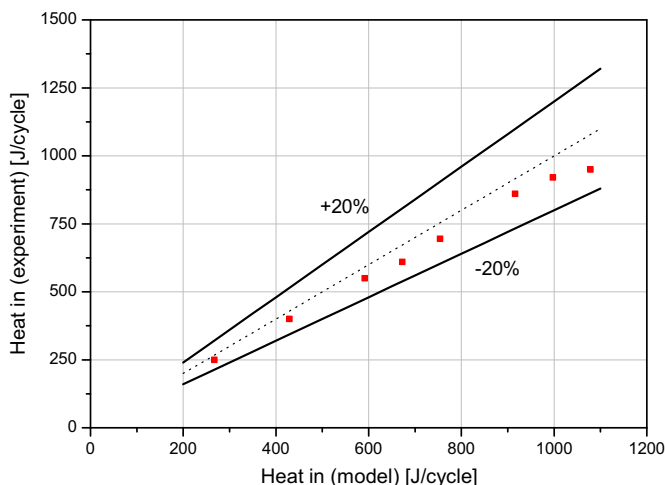
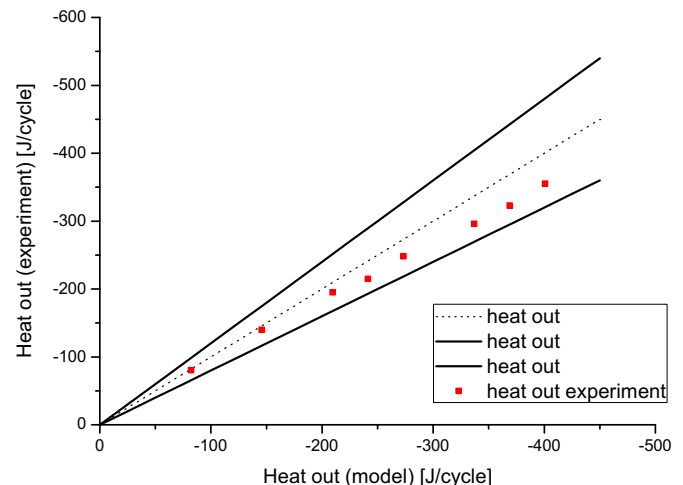
The deviation observed in the power ratio (Fig. 14) also exhibits low variance across the entire engine operation spectrum. Once again modeled results are improved in comparison with the experimental values, with an average deviation of about 9%. In addition to the probable causes mentioned above, the motor efficiency should also be investigated in the specific case since the efficiency of these parts typically varies between 90% and 98%. A slip in the drive system between the engine crankshaft and the

motor shaft should be considered as well. The Solo Stirling V161 cogeneration unit showed significantly very good performance compared to corresponding thermal power CHP systems of internal combustion engine (Table 9). The Stirling engine CHP units are more compatible to internal combustion engine CHP units, in terms of their capacity, the fuel type used, the installation and maintenance procedure. In the present study, the thermodynamic analysis and performance of the Solo Stirling engine is tested and compared with relevant published data for similar Stirling and internal combustion engines. Further study of these systems in terms of installation and operating costs, unit reliability and emitted pollutants would be of interest.

The efficiency of the cogeneration unit is expressed by the ratio of primary energy savings PESR. Using the efficiency level reference values as defined by [8]:

$$\text{PESR} = 1 - \frac{1}{\frac{n_e}{n_{er}} + \frac{n_h}{n_{hr}}} \quad (4)$$

The reference efficiency value for separate thermal production and units manufactured after 2006 was determined as $\eta_{hr} = 0.525$ which is based on the lower heating value capacity of the fuel and standard ISO conditions (ambient temperature 15 °C, pressure of 1.013 bar, 60% relative humidity) [8]. Due to the fact that the unit has been installed in an area in which the average annual temperature and relative humidity values differ from the standard ones, the use of correction coefficients is imperative. According to Ministerial Decision B-1490-2009 [9], for produced voltage of <0.4 kV, the correction coefficient for the electrical power supplied to the grid is 0.860 and for the electrical power consumed by the unit is 0.925. Thus:

**Fig. 10.** Heat input experiment–model relation.**Fig. 11.** Heat output experiment–model relation.

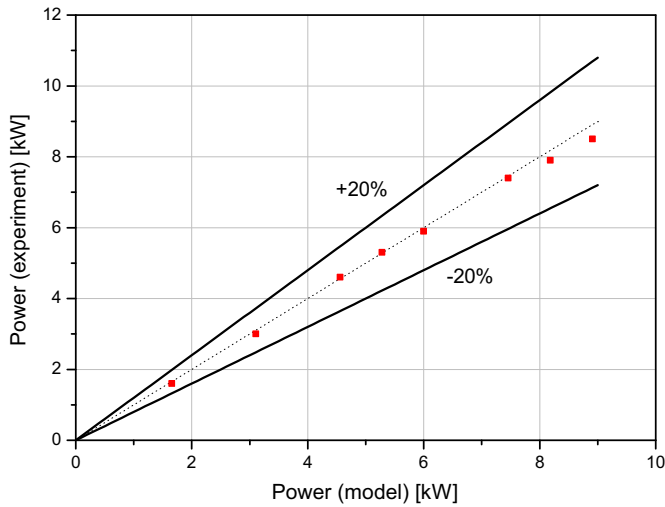


Fig. 12. Power experiment–model relation.

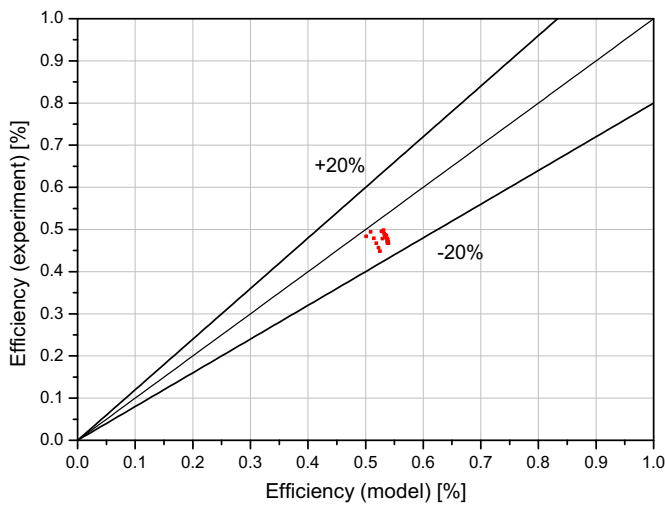


Fig. 13. Efficiency experiment–model relation.

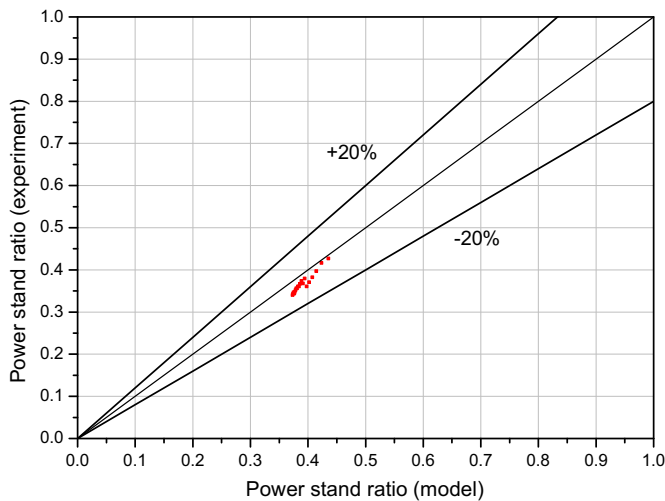


Fig. 14. Power stand ratio experiment–model relation.

$$n_{er} = 52.5\% \cdot (0.860 \cdot 15\% + 0.925 \cdot 85\%) = 48.1\%$$

The primary energy saving ratio (max. load):

$$PESR = 1 - \frac{1}{\frac{n_e}{n_{er}} + \frac{n_h}{n_{hr}}} = 0.368 = 36.8\%$$

6. Conclusions

The aim of the current work was to present the installation, explain the operation and analyze the results of the thermal performance of the CHP Solo Stirling Engine V161 unit at the Laboratory of Applied Thermodynamics of the National Technical University of Athens. For this purpose, a computer code was developed, based on the adiabatic model, for the thermodynamic analysis of the Stirling engine. Heat balance analysis was applied to the entire operating range of the unit and the performance of the Solo Stirling V161 unit was compared with other CHP systems of corresponding performance. Furthermore, the energy savings achieved through the use of the Solo Stirling V161 unit were compared to relevant results of other power systems intended for co-producing electricity and thermal energy. In summary, it was observed that:

- A significant mass of the working gas remained continuously in the cooler volume so it could be classified as “inactive mass”. As a result, reduction of the heat transferred and thus, the efficiency of the engine were observed. Similar conclusions were reached for the regenerator and for the heater volumes.
- The generated electrical power increased almost linearly compared to the load of the unit.
- The electrical efficiency was approximately constant for mean pressures of 70–130 bar (50–100% of the engine’s load).
- The thermal efficiency of the unit decreased slightly once the mean pressure of the engine increased (load).
- The overall efficiency of the unit suffered small changes throughout the operating load range which varied from 88.84% to 92%.
- The electrical to thermal power standard ratio of the Solo Stirling V161 was calculated at $C = 0.35$.
- The primary energy savings ratio (max. load) was calculated $PESR = 36.8\%$.

Nomenclature

g_{Ah}	mass flow rate in the heater. [kg/sec]
g_{Ak}	mass flow rate in the cooler. [kg/sec]
g_{Ar}	mass flow rate in the regenerator. [kg/sec]
n	Overall efficiency. [%]
n_e	electrical efficiency. [%]
n_h	electrical efficiency. [%]
P	the power output. [W]
P_e	the electrical derived power from the cogeneration unit. [W]
Q_a	ventilation heat energy loss. [W]
Q_c	exhaust gas heat energy loss. [W]
Q_h	the heat absorbed by the heater. [J/cycle]
Q_k	the heat rejected by the cooler. [J/cycle]
Q_r	the remained into the regenerator metal matrix. [J/cycle]
Q_u	incoming energy rate. [W]
Q_w	external cooling circuit heat energy loss. [W]
T_k	cooler gas temperature. [°K]
T_h	heater gas temperature. [°K]
W	the net produced work by the engine per cycle. [J/cycle]

W_c the produced work by the power piston per cycle. [J/cycle]
 W_e the consumed work by the displacer piston per cycle. [J/cycle]

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